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Department of the Navy
Bureau of Ordnance
Contract NOrd 9612 - Task 4

CAVITATION IN AXIAL FLOW PUMPS

TERMINATION REPORT

Hydrodynamics Laboratory
California Institute of Technology
Pasadena, California

Report No. E-12.18
January 1956
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ABSTRACT

A series of cavitation experiments in an axial flow pump are described. It is found that cavitation first occurs in a tip vortex formed by the flow through the tip clearance space of the rotor blade. The effect of tip clearance and several cavitation suppression devices are shown. The schemes tested were rigid end plates attached to the blade ends, strings, and a continuous rotor shroud. Of these, the shroud appears to delay the inception of tip cavitation best for axial flow pumps.

I. INTRODUCTION

Summary of Previous Work

In the past, propellers have been used for the propulsion of underwater missiles and surface craft. As successful methods of acoustic detection were developed it was found that the noise produced by the propeller was excessive for both active and passive search. The problem has become more severe with the continued emphasis on high speed. It is principally for this reason that the shrouded propeller or pumpjet has been introduced. The potential improvement offered by the pumpjet is due to the increase in static pressure that can be provided near the blade row. The pump can then be so designed as to operate at any forward speed with no blade cavitation, and the limiting speed is determined by cavitation on the exterior parts of the shroud. Unfortunately, it was soon found that cavitation occurs near the blade tips in a pump long before surface pressure calculations indicate that it should. The inherent advantage of the pumpjet is partially offset by such premature cavitation and noise generation.

It was therefore decided to study cavitation in the tip regions of the blades of an axial flow pump of a type suitable for a pumpjet. The principal objectives were to determine the cause, location and possible methods of eliminating premature cavitation noise. Accordingly, an experimental facility was constructed that permitted the direct visual observation of the flow around the rotor blades, as well as measurements of the velocity profiles, and so forth, between the blade rows. The design and construction of the axial flow pump test facility was reported in Ref. 1, and will not be further elaborated upon herein. Figure 1 is a photograph of the test section and working area of the pump facility showing the window of the test section through which the blades of the pump can be viewed.

Very early in the course of working with the pump it was discovered that cavitation occurred in and near the gap formed by the clearance between the rotating blade and the stationary case. Figure 2 shows the cavitation in this region. Cavitation in the gap itself can be eliminated by proper rounding of the leading edge or pressure surface of the blade.(2) The remaining cavitation in Fig. 2 occurs in a vortex and is termed herein as "tip vortex" cavitation. Such cavitation has many similarities to the tip vortex observed on propellers. However, the details leading up to vortex cavitation in a pump are somewhat different from those in a propeller operating in a free stream.

The first investigations of the project were concerned with the determination of the inception cavitation number K_i* for the blades of the pump described in Ref. 1 as a function of the tip clearance and flow rate coefficient.(2) The inception point was determined visually using a stroboscopic light to observe the cavitation bubbles. It was found that there was an "optimum" tip clearance for the blades which (at a given flow rate coefficient) gave a minimum value for the inception cavitation number. The values of K_i increased gradually with increased tip clearance but increased sharply for very small clearances. For comparative purposes similar experiments were conducted on a hydrofoil in the High Speed Water Tunnel at the California Institute of Technology. The hydrofoil was set up with variable tip clearance next to the window of the working section. Tip vortex cavitation was also observed, although it was not as well defined as comparable cavitation in the pump. The behavior of the inception cavitation number with tip clearance was similar to that in the pump for large clearances, but it did not exhibit a minimum at smaller clearances. This difference was ascribed to the "scraping effect" of the rotating blade in the pump. Tip vortex cavitation on a stationary hydrofoil appears to be important, although not as severe as on a blade moving past a stationary wall. This effect will be mentioned again in a later discussion.

Since it is not cavitation, as such, that is of interest, but rather the noise that cavitation generates, considerable effort was expended to determine the inception of noise (or cavitation) by direct acoustic measurement. For this purpose the case of the pump had been drilled and fitted with a barium titanate pressure transducer. The opening into the machine was 3/16-in. diameter and was carefully filled with petroleum jelly to exclude air. One tap was installed immediately in front of the rotor row and another was directly behind the blade row. In this way, the sound receiver was very close to the origin of the noise and a high signal to background ratio was achieved. The measurement of the signal output posed

*See nomenclature for definition of symbols.

a problem since the sources of disturbance were spaced equally around the blade row and rotated at the angular speed of the pump. In order to measure the peak sound pressure, rather than an average value, the output was read directly on an oscilloscope screen. In Ref. 3 the acoustic inception of cavitation was defined as that cavitation number which gave noise peaks on about 75 per cent of the blades in the rotor row. The incipient cavitation data so determined compared fairly well with the visual determination (to within 15-20 per cent) and exhibited the optimum observed before. However, this definition of inception is somewhat arbitrary and an improved method was subsequently used. Figure 3 shows the results of the visual and acoustic measurements of the inception point as defined above.

The relative increase in noise intensity as a function of cavitation number was measured and is shown in Fig. 4. It is apparent from these curves that noise occurred long before the "inception point", as defined previously. It was thought, however, that the manner in which the inception data were taken was somewhat more representative of the blade design than the first appearance of noise on an individual blade.

It was only natural at this juncture to make some attempts at noise reduction in the machine. The approach was motivated by the fact that cavitation occurred first in the tip vortex, so that cavitation would be delayed if the vortex were to be destroyed. At that time it appeared possible to isolate the noise performance of each of the sixteen blades in the rotor row by use of the oscilloscope and synchronized sweep technique. Accordingly, various tip cavitation suppression devices were attached to groups of about four rotor blades. It was soon found, however, that the blade attachments sufficiently disturbed the flow at the rotor tips to cause significant peripheral nonuniformities. It was not possible then to compare strictly the blades with attachments to those without. However, preliminary results (Ref. 4) indicated that some of the vortex suppression schemes showed promise. It should be mentioned here that calculations of the blade surface pressure coefficient at the design flow rate indicated that cavitation should occur at $K_i = 0.66$, whereas acoustic measurements gave an incipient cavitation index of about 1.12[4] for bare blades with "optimum" clearance. It can be seen that there is room for considerable improvement in reducing the incipient cavitation number.

The basic notions regarding the formation and location of the tip vortex are contained in Refs. 2, 3 and 4. A rough theory that accounts for the presence of the vortex is outlined in Ref. 4. In this theory, the flow through the tip clearance space is considered to be bounded by two vortex sheets, one sheet springing from the blade end and the other sheet being an image of the first sheet in the wall. The vortex strength of the sheet is supposed equal to the local value of the vorticity at the point where the sheet originates along the chord of the blade tip. The vortex sheets, so determined, give rise to the jet of liquid that flows through the clearance space. The vortex sheet is not stable, and as a consequence it immediately rolls up into a tube. Due to molecular and turbulent mixing processes, the vorticity contained within the surface of the tube is diffused over a small region, not unlike that of a Rankine combine vortex. The spinning action of the motion lowers the pressure at the center of the vortex core so that cavitation can occur there first.

This theory seems to be qualitatively verified by the cavitation photographs of Ref. 2, one of which is shown in Fig. 2. An estimate of the inception cavitation number was made using this approach and compared to experiment in Ref. 4. The tip vortex theory gave a value of $K = 1.05$ for inception, whereas the acoustic measurement quoted above gave $K = 1.12 - 1.18$. The agreement, although not perfect, bears out the general premises of the theory.

Subsequent to the preliminary observations and measurements given in Refs. 1 - 4, additional attempts were made to suppress the cavitation noise. The remainder of this report describes these experiments and their results, and discusses some of the experimental difficulties associated with cavitation noise experiments in a rotating machine.

II. EXPERIMENTAL WORK

1. Noise Suppression Devices

Several techniques have been suggested for suppression of the tip vortex cavitation noise. Most schemes involve the attachment of some device to the blade to destroy the tip vortex. Among these are: attachment of strings to the blade, attachment of flexible end plates to the blade ends, attachment of rigid end plates and fastening a continuous shroud to the blade row. As mentioned before, it was found necessary to make each blade identical insofar as possible to secure uniform flow conditions around the rotor periphery. Experiments were then made with identical strings attached to each of the sixteen rotor blades, identical end plates, and so forth. However, it was concluded from preliminary work that the flexible end plates were unsatisfactory and they were not further studied. Figure 6 shows a rotor blade with a rigid, lucite end plate attached, and Fig. 7 shows the installation of the continuous lucite rotor shroud. The shroud was about 1/4-in. thick and the blades were fitted into slots recessed into the shroud. The shroud was made in this fashion since it was not desired to remachine the pump casing. The flow was carefully faired onto and away from the shroud by contoured rings attached to the case. The radial clearance between the shroud and case was 0.015 in. The space between the blade and the slot was carefully filled and smoothed with wax before operation.

The string attachment is shown in Fig. 8. The string is of nylon and is cemented into a slot near the leading edge. In general, it was found that the disturbance caused by the attachment caused cavitation to occur there. The present method, however, is free of this difficulty.

2. Instrumentation and Experimental Procedure

The noise produced by the cavitation was detected by a barium titanate crystal hydrophone. The output of the hydrophone was amplified and viewed on the screen of an oscilloscope. An attenuator was also introduced into the circuit to facilitate measurement of the output. The cavitation noise output was found to be very sporadic and intermittent. In order to reduce errors of human judgment and fatigue, a time exposure photograph was made of the oscilloscope screen. The camera pictures then served as the primary data for the measurement of cavitation noise. These data, together with the attenuator readings, permitted the computation of noise output in db over the noncavitating background.

All noise measurements were made in the frequency range of 20-100 kc. Machinery noise was not found for frequencies much more than about 10 kc, so that it was not included in the noise measurements. The procedure was then as follows: with the machine running with no cavitation, the noise background level was arbitrarily set with zero db on the attenuator and with a scope gain to give a band or deflection on the screen of about 1/8 in. When cavitation commenced, the attenuator was adjusted to keep the scope deflection on the screen. The screen pattern was photographed with the Land camera using an exposure of 30 sec. The db rise was then computed from the measured peak height on the picture, the known background height and the attenuator reading. The noise-measuring equipment is shown in Fig. 9, in which the oscilloscope, hydrophone and filtering equipment can be seen.

The present test facility is limited to rather low rotative speeds by the blade design and available power. For this reason it was necessary to make the ambient pressure in the working section of the pump low (about 5-15 ft of water absolute) in order to obtain the desired cavitation numbers. As a result, the entire piping system was under a vacuum. At these pressures a considerable amount of dissolved air had to be removed so that free air bubbles would not recirculate through the test section. It was found that the presence of small undissolved gas bubbles gave an erroneous noise indication, so that no air leaks whatsoever could be permitted in any part of the circuit. In fact, the occurrence of minute air leaks caused considerable experimental difficulty and loss of time. To prevent air leakage with low absolute pressures it was found necessary to flood all joints and flanges with water where possible, and to coat the remaining flanges with neoprene paint with the system under vacuum. It was possible to eliminate all air leakage with this technique. The dissolved air content was kept between 5-6.5 parts per million (ppm) by weight, as determined by the Van Slyke apparatus. The volume of the circuit is 1200 gallons, and to obtain these comparatively low air content values for this amount of water it was necessary to operate the system under the maximum vacuum available (about 27 in. Hg. at the high point in the circuit) for periods up to thirty hours.

In the first experiments (those quoted in Refs. 1-4 and Figs. 3-5) the rotative speed was 600 rpm, corresponding to a tip speed of 36.5 ft/sec. After prolonged operation at this speed it was found that several entrance blades had failed in bending. The blades were successfully repaired, but in all subsequent operations the rotative speed was maintained at 500 rpm and no failures since that time have been experienced.

The pump can accommodate a maximum of three complete stages plus two rows of straightening vanes. For the cavitation tests only one stage was used, i.e., an entrance row, a rotor row and a stator row. The basic stage design is repeating, that is, the flow leaving the stator blades is the same as that leaving the entrance blades. Consequently, there is a certain amount of whirl remaining in the flow leaving the stator row. For the cavitation tests, straightening blades were not used and the whirl energy was not recovered. The stage was "spread out" for these tests, so that the interference between the entrance and rotor rows was minimized and the cavitation made more easily observable. The chord of the entrance, stator and rotor blades is about 2 in. When the rotor row is put in the second stage position it is 5-1/4 in. downstream of the entrance blades and the stators were mounted 5-1/4 in. downstream of the rotor row.

In all these experiments the inception and noise data were taken with <u>decreasing</u> cavitation number rather than with increasing K as was done by Shapiro[5] and Parkin.[6] This procedure was followed for two reasons: first, the pressure control equipment was such that the pressure could not be conveniently increased; and secondly, it was believed that the inception phenomena characteristic of decreasing K would be more typical of actual field conditions than increasing K. This point will be mentioned again in the discussion.

3. Presentation of Results

The experimental results not reported before consist of tests on the rotor row completely equipped with string attachments, rigid end plates, continuous shroud , and additional experiments on the bare blades. The tip clearance was maintained from 0.015 to 0.026 in. for all devices. In order to check on the reproducibility of the data, the tests were re-run various times and not always with the same results, as will be seen. The chronological order of the tests was as follows:

i. Preliminary experiments at 600 rpm, including plain blade experiments, effect of tip clearance, partial shroud, strings on individual blades, etc. (reported in Refs. 1-4).

ii. Tests on rotor row completely equipped with lucite end plates.

iii. Tests on rotor row completely equipped with string attachments.

iv. Tests on rotor row completely equipped with lucite end plates.

v. Tests on rotor row with plain blades with 0.026-in. tip clearance.

vi. Tests on rotor row equipped with continuous lucite shroud ring.

vii. Tests on rotor row with plain blades with 0.015-in. tip clearance.

viii. Efficiency data for the shroud ring.

Items iii to viii were run at 500 rpm, and all others at 600 rpm. The results of the cavitation experiments with lucite foot attachments are given in Fig. 10. Data for the string attachment is given in Fig. 11 and for the continuous shroud in Fig. 12. Additional results on the plain blades (items v and vii) are given in Fig. 13. Finally, some measurements of stage efficiency with and without the rotor shroud are summarized in Fig. 14.

All of the tests except those of Items vi and viii were conducted with the full rotor diameter (14.00 in.). The presence of the shroud reduced the effective rotor diameter to 13.48 in. Although the blade shape is effectively the same, it was necessary to account for the shroud thickness in determination of the flow coefficient ϕ and cavitation number K. It should also be mentioned that in analyzing the data of Refs. 2, 3, and 4, it was found that the cavitation numbers were too high by a factor of 1.085. This error was corrected in the reproduction of Fig. 5 of this report.

III. DISCUSSION

Plain Blades

From Figs. 5 and 13 it is apparent that the general observations of Ref. 4 concerning an optimum tip clearance are borne out, i.e., an optimum at a value λ (tip clearance divided by maximum blade thickness) about 0.10. There is, however, a systematic difference between the tests at 600 rpm (Fig. 5) and those at 500 rpm (Fig. 13). This difference in inception cavitation index amounts to about 12-13 per cent and the lower speed tests give lower values of K. The amount of dissolved air is very nearly the same in both tests (between 6 and 6.7 ppm). This difference is thought to arise from the Reynolds number scaling effect observed by Kermeen(7) in which it is found that inception cavitation numbers increase with Reynolds number. Thus tests with low Reynolds numbers (or low speed) give optimistic results. Although there is not a large speed change (20 per cent) the direction and amount of change corresponds to that found in Ref. 7 on bluff bodies. The scatter in individual runs under the best conditions amounts to about 6 - 7 per cent in inception cavitation number, so that this effect is not believed responsible for the difference.

The very first appearance of visual cavitation is also noted on these figures. It can be seen that visual cavitation usually occurs significantly below the inception point. For these tests, the difference in inception cavitation number determined visually and acoustically amounted to a maximum of 18 per cent and averaged around 10 per cent. No particular significance is ascribed to this difference. It is believed that with improved visual techniques and sufficient care, there would be no discrepancy in visual and acoustic inception measurements. It should be mentioned that a small amount of debris on the blade leading edge, or a small amount of undissolved air in the water, significantly changed these results. Great care had to be exercised to obtain even reasonably repeatable and reliable data.

The effect of a small amount of undissolved air in the water is shown in Fig. 4, in which it is seen that the noise intensity starts at zero db and grows uniformly as K is decreased. However, when such air is scrupulously avoided in the working section, inception is marked by discontinuous jump in noise output of about 5-20 db. In fact, all of the present tests that are considered reliable showed such a jump. This observation may not apply to other machines or experimental situations.

Tip cavitation when it occurred in the pump was extremely sporadic. At or near the inception point, cavitation might occur only once every 1- 5 sec. It was for this reason that a 30-sec. exposure was made to record this phenomenon. Slow speed motion pictures taken of tip cavitation with a string suppressor near the inception point showed bubbles appearing only infrequently, perhaps one in 10-15 frames. This points out the fact that the flow field in the tip vortex is highly irregular and fluctuating, a consequence that makes the visual inception of cavitation difficult to determine.

The maximum noise output (with reference to the noncavitating background in the 20-100 k.c. range) was about 40-50 db. In most tests, however, no great effort was made to find the maximum. It would have been of great interest to refer these data to free field conditions. Unfortunately, for the present experimental arrangement this is a practical impossibility. This circumstance is particularly regrettable since it might be inferred from Robertson's article[8] that tip vortex cavitation in a pumpjet is "small". The writer questions this result for several reasons: (1) In his tests, the total integrated sound output of the entire tip vortex was not evaluated at free field conditions; (2) the use of a narrow band detection apparatus in a water tunnel may lead to spurious results; and (3) no mention is made of air content or scaling effects of such tip vortex cavitation. Furthermore, the experiments of Ref. 8 were conducted on a high aspect ratio elliptical wing, and it is certain that the structure of the tip vortex is not the same as that of a hydrofoil set up with a small end or tip clearance as in the case of a deflected control surface or a moving vane as in a pump.

In view of the later and more refined inception measurements given in Fig. 13, the theory put forward in Ref. 4 appears to give a better prediction than at first thought. Consequently, this theory should serve as a useful indication of the increment in inception cavitation number over the blade surface calculation.

<u>End Plates</u>

The end plates shown in Fig. 6, when fixed to the rotor tips, prevented the occurrence of tip vortex cavitation. The plate was 2.5 in. long by 0.6 in. wide and 0.055 in. thick. All edges were carefully faired to prevent local cavitation. The results shown in Fig. 10 are somewhat different from each other, as before the 600 rpm values being highest. It is believed that the pressure forces on the "foot" at 600 rpm are sufficiently high to cause bending and premature cavitation at the leading edge of the foot, if not actual scraping on the case. The 600 rpm values are not considered reliable for this reason. The tip clearance ratio (again figured on the maximum blade thickness) was 0.178 and had inception values of $K_i \approx 0.90 - 1.05$. The comparable plain blade data are for $\lambda = 0.171$ and give $K_i = 0.95$, all values being for the acoustic inception cavitation number. Because of the weak strength of the end plates in bending, it is not considered a practical device. In any event, the improvement is small.

String Attachments

String attachments have received considerable attention as tip cavitation suppression devices. The strings used for these tests were of nylon, about 2.5 in. long and 1/32 in. in diameter. At the outset, it was noted that the string was subject to the irregular fluctuations in and near the tip vortex, so that it did not always lie in the vortex. The attachment point and length of string were critical factors in determining the position of the string in the flow. It is probably for this reason that the string attachment shows no improvement over the plain blade (Fig. 11), and is, in fact, somewhat worse.

These results are in contrast to the results of Robertson(8) and Thurston(9) who find strings to be useful inhibitors of tip vortex cavitation noise. In both cases, these results were obtained on propellers, not pumpjets. The tip vortex that occurs in the wake of a propeller operating in a free stream arises from the roll-up of the vortex sheet that springs from the trailing edge of the blade. This rolling-up process is fairly slow and may take several chord lengths to become reasonably complete. The situation is quite different in a pumpjet with rotor blade tip clearance or a control surface with end gap. In these cases it is shown in Ref. 4 that the vortex sheet springs from the end of the blade and is distributed along the chord. The resulting roll-up is more rapid and since the clearance is usually quite small the vortex is concentrated and, therefore, intense.

For this reason it is difficult to destroy with a string device. It must also be pointed out again that all of the pump tests were conducted with a set of entrance blades, the wakes of which contribute unsteadiness of the rotor flow.

In view of these facts, no great hope is held out for string cavitation suppression devices in pump jets, although they are useful for propellers.

Shrouded Rotor

The shrouded rotor appears to give the best performance of all devices with a minimum K_i of 0.83 and a maximum of $K_i = 0.90$. These results are, on the average, about 10 per cent lower than the optimum cavitation inception number observed on the plain blades. It should be noted that if the visual appearance of cavitation is taken as the criterion of cavitation inception, then the shrouded rotor is even more favorable, being about 18 per cent lower than the plain blades. The clearance between the shroud and stationary case was 0.015 in. corresponding to a value of $\lambda = 0.10$. There is no doubt that further improvement would result if the running clearance (or the resulting leakage flow) were to be reduced.

When the rotor is shrouded, the leakage from the high pressure side through the clearance space to the low pressure region at the inlet to the blade row penetrates into the main stream and creates a large disturbance there. This was evident in the present tests, since cavitation occurred first on the blade surface about 1/8 to 1/4 in. inboard from the rotor tip. An only partially successful attempt to seal the clearance was made by fastening a rubber strip onto the shroud in such a way that the pressure difference across the rotor would seal the gap. In operation, the rubber strip fluttered and made some noise itself. Although the results were inconclusive, some promise was shown. Of course, in any actual machine considerably more sophisticated sealing arrangements could be made.

The effect of the leakage and the additional skin friction of the shroud change the performance and cause a slight reduction in efficiency of the stage. The stage performance with and without the shroud is shown in Fig. 14, in which it is seen that maximum efficiency is reduced about 5 - 6 per cent. Both the head and torque are seen to increase with the presence of the shroud. Part of the torque increase is due to the added skin friction. However, the remainder must be attributed to the slight change in rotor geometry and to the different effects of the tip clearance flows in the two cases.

Effect of Design

In all of the experiments reported thus far, the pump has been operated at its design rate of flow or flow coefficient of $\phi = 0.45$. This value corresponds to an angle of attack of 1.5 degrees and is typical of well-designed axial flow compressors. The angle of attack is measured between the blade chord line and the vector mean of the inlet and outlet velocities. All tests showed that operation at higher flow rate coefficients or lower angles of attack decreased the inception cavitation number. The order of magnitude of the change is given in Fig. 3. There is no doubt that from the standpoint of cavitation, a better design could be made than the one used. However, the purpose of these tests was to investigate tip vortex cavitation, and their value is in the comparison of the vortex cavitation to that with the suppression devices and theoretical surface pressure calculations for a given set of conditions. For this reason an extensive investigation of tip cavitation for all flow rates was not undertaken.

If a pumpjet were to be designed for maximum cavitation resistance the following rules should be observed. Entrance blades or thick struts upstream of the rotor should not be used. The blades should be as thin as practical (say 6 - 7 per cent) to reduce the surface overspeed velocities. The lift coefficient of the tip should be lower than typical compressor practice calls for (say 0.4). This may necessitate the use of higher solidity than is customary and, finally, the blades should be operated at the "smooth entry" condition. For normal designs, this corresponds closely to zero angle of attack.

IV. SUMMARY AND RECOMMENDATIONS

An experimental study of cavitation inception has been conducted in an axial flow pump. It was determined that cavitation first starts in a tip vortex formed by the flow through the tip clearance gap. An optimum tip clearance (ratio of gap to maximum blade thickness) was found to be about 0.10

for the design studied. At this condition the inception cavitation number was determined acoustically and found to be about 39 per cent higher than that computed on the basis of blade surface pressure coefficient. An approximate theory of tip vortex cavitation inception agreed to within a few per cent of the experiments, indicating that it is useful for preliminary calculations. Various noise suppression devices were tried, among them being strings, end plates and a shroud. Of these, the only one showing an appreciable gain was the shroud.

In the design of quiet-running pump jets, it is recommended that no entrance blades be used and that the rotor be operated at zero angle of attack.

The limitations of the experimental facility prevented the investigation of an important sound suppression technique, i.e., injection of air into the tip vortex. Although the air bubbles would not be absolutely quiet, it is believed that a significant reduction in inception noise could be made with air injection, and it might become possible to operate at cavitation numbers much lower than incipient with an acceptable noise magnitude. It is suggested, therefore, that air injection experiments be made on hydrofoils set up with a tip clearance in a water tunnel where the air content can be controlled and a certain amount of free air can be redissolved in the circuit.

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NOMENCLATURE

A - cross-sectional area of pump annulus

D - diameter

g - gravitational constant

h - pump head (ft)

K - cavitation number $= p_\infty - p_v / \rho w_\infty{}^2/2$

p - pressure

Q - flow rate (cfs)

T - torque

U - peripheral velocity $= \dfrac{D\omega}{2}$

w - velocity relative to blade

α - angle of attack (measured between blade chord and mean velocity w_∞)

ρ - density

ω - angular speed

ϕ - flow rate coefficient $= \dfrac{Q}{A\ U_2}$

ψ - head coefficient $= h/U_2^2/g$

τ - torque coefficient $= \dfrac{T}{\rho \dfrac{AD_2 U_2{}^2}{4}}$

η - efficiency $= \dfrac{\phi\,\psi}{\tau}$

λ - tip clearance ratio = gap/maximum blade thickness

SUBSCRIPTS

∞ - refers to vector mean of inlet and outlet velocity

v - vapor pressure

2 - tip conditions

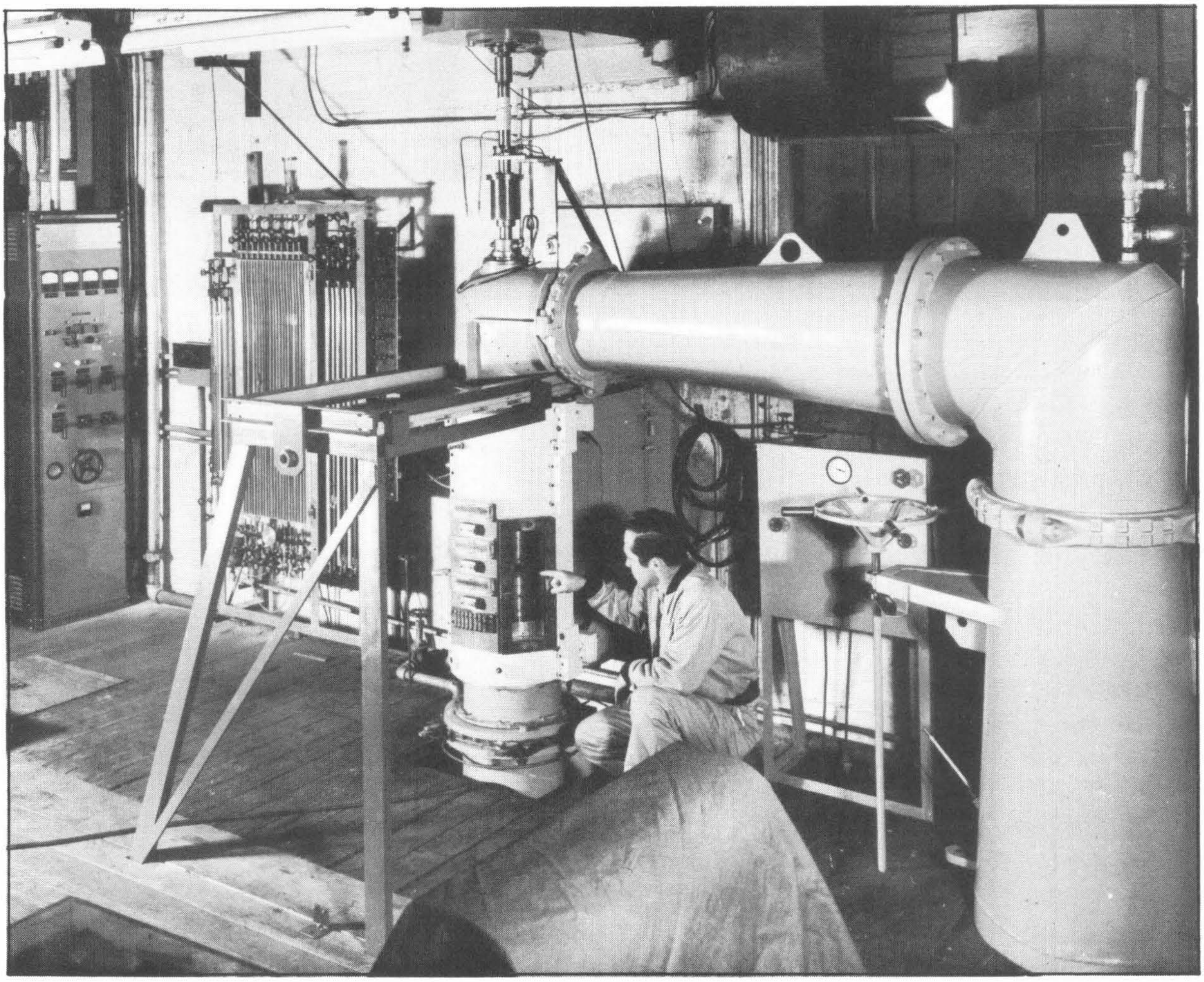

Fig. 1 - Axial flow pump and working area.

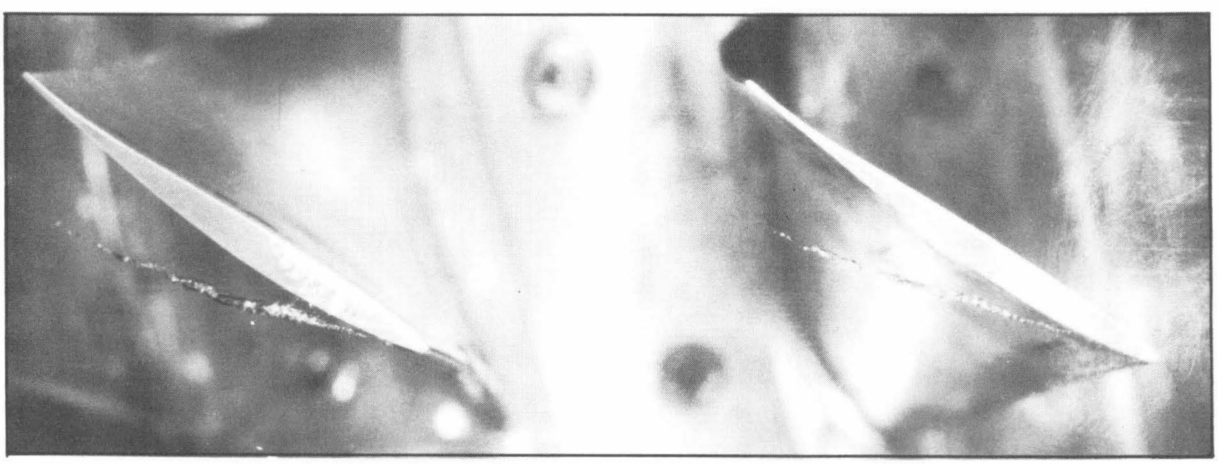

Fig. 2 - Tip vortex cavitation and cavitation in the tip clearance space. The pressure side edge of the right-hand blade has a 0.02 in. radius, and the tip clearance ratio is $\lambda \doteq 0.13$.

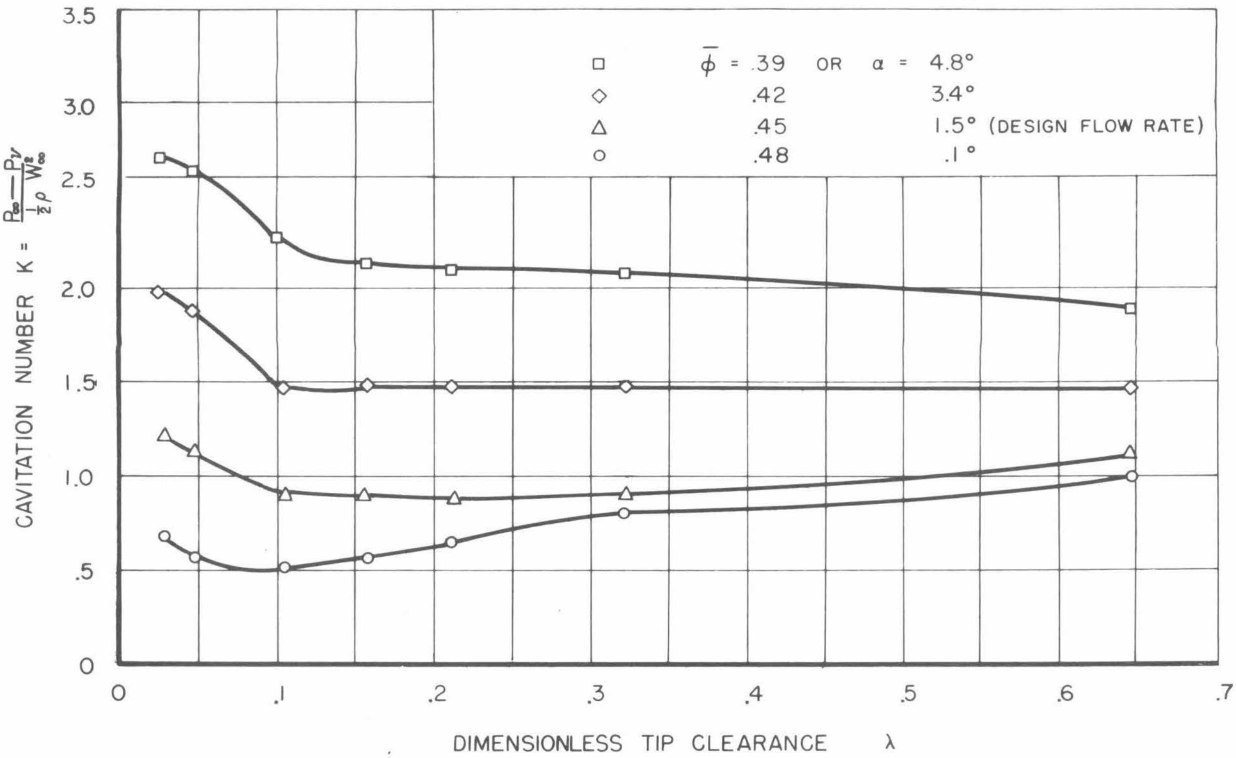

Fig. 3 - Incipient cavitation at design flow rate ($\phi = 0.45$) for various tip clearances determined visually (Ref. 2).

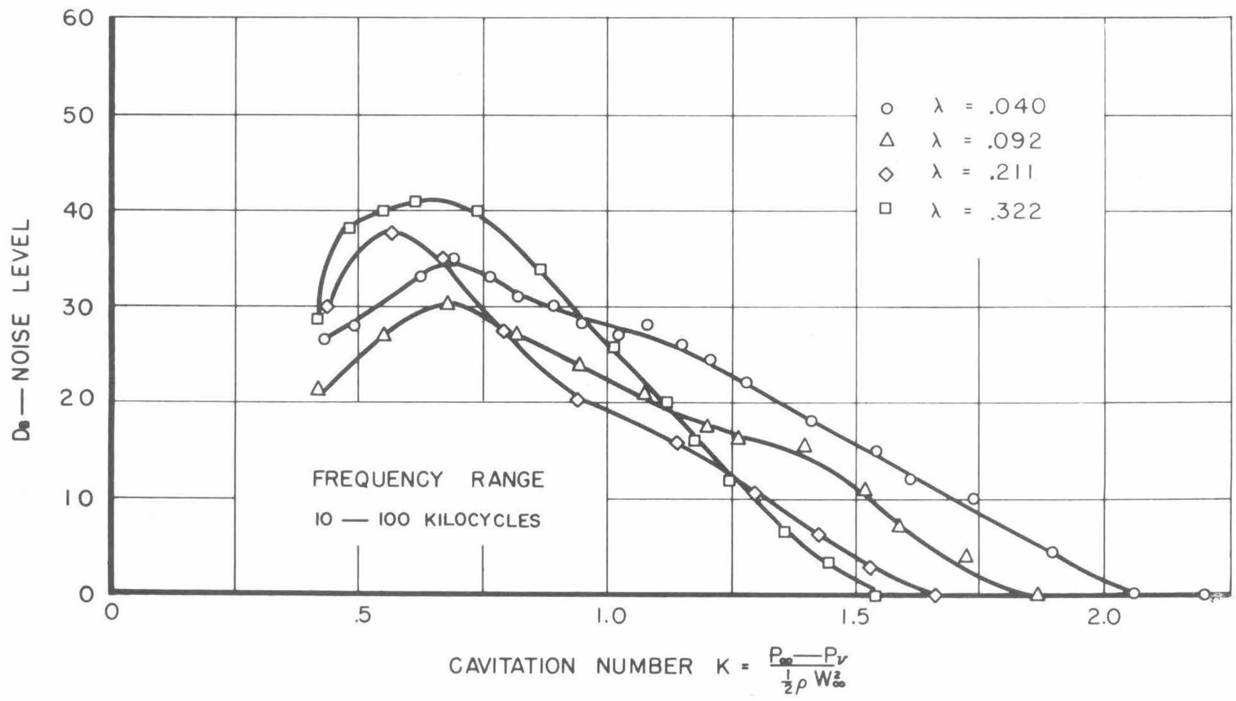

Fig. 4 - Noise level as a function of cavitation number for several tip clearances (Ref. 3). (The cavitation numbers are too high by a factor of 1.085).

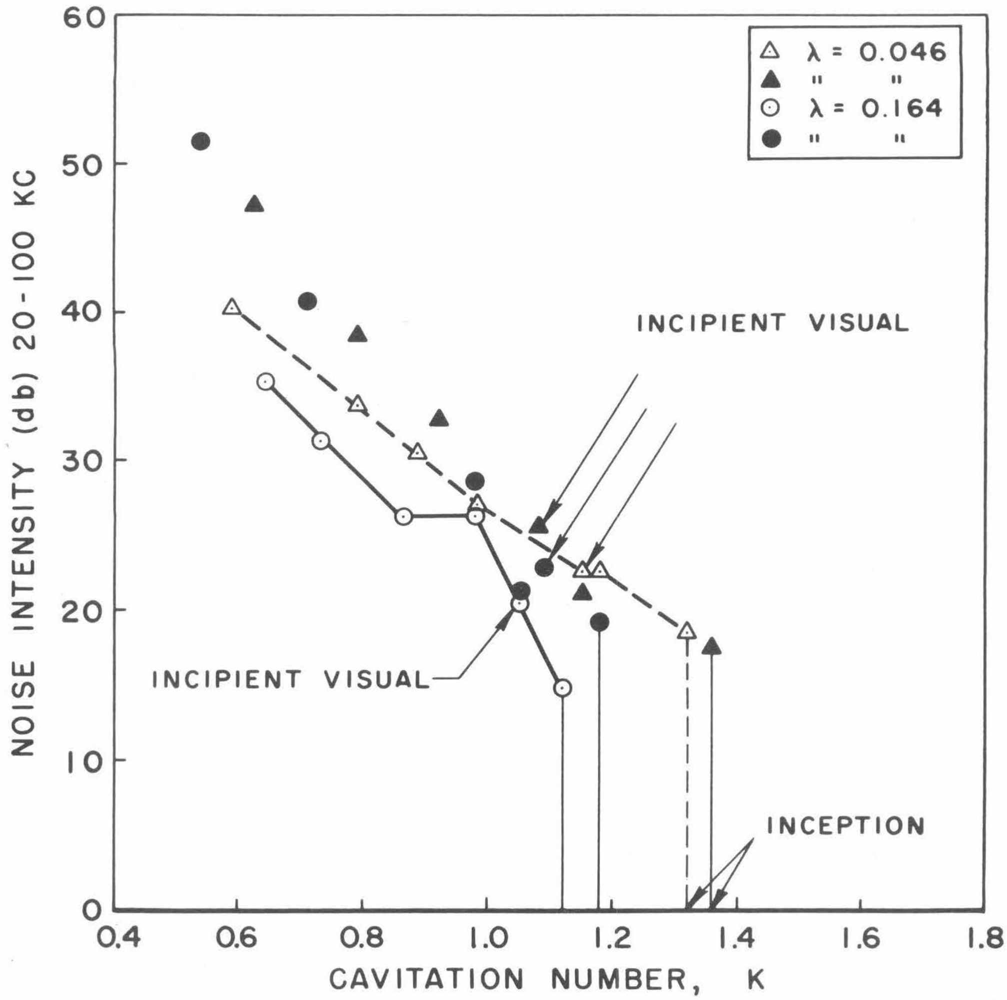

Fig. 5 - Noise level as a function of cavitation number for plain blades at the design rate of flow for two tip clearances (600 rpm).

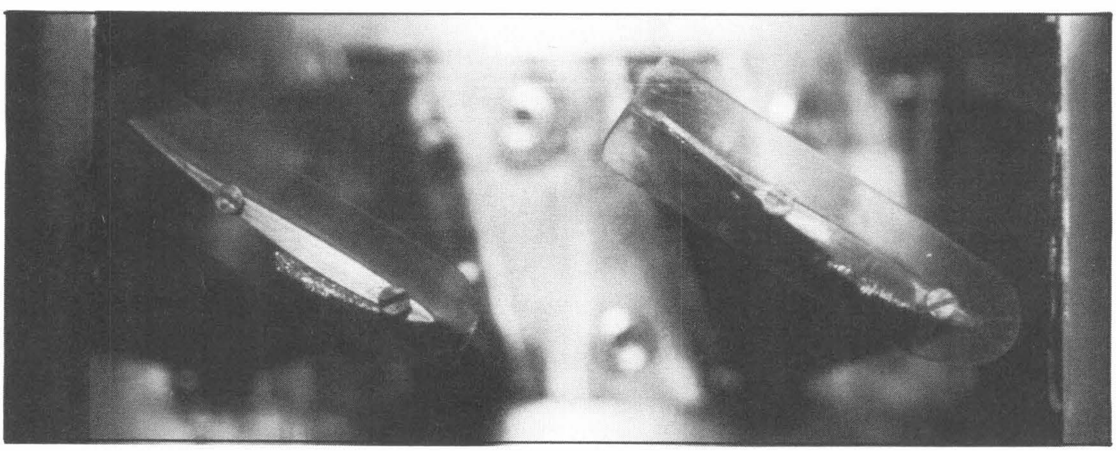

Fig. 6 - Rotor blade, with lucite end plate attached, showing extensive cavitation at the design flow rate, $K = 0.56$, $\lambda = 0.164$.

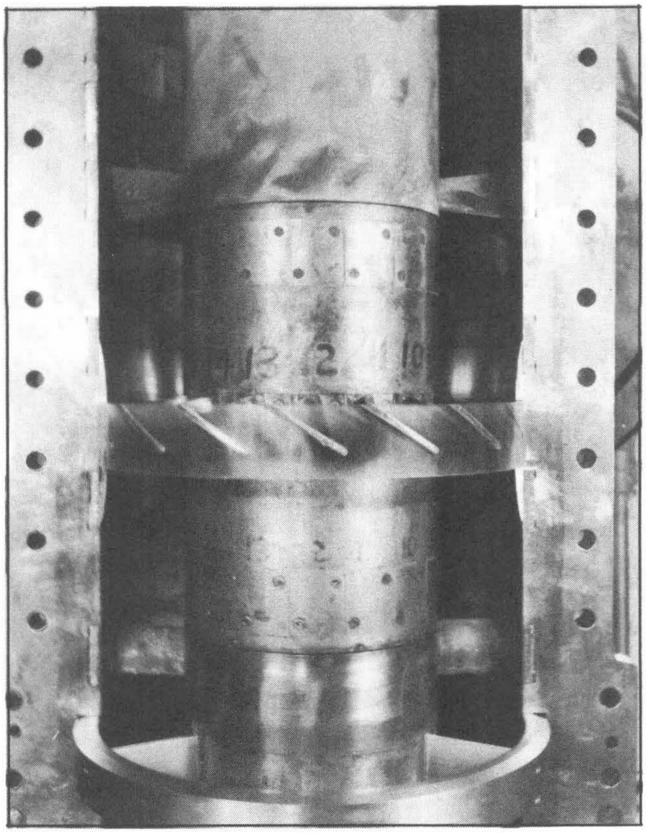

Fig. 7 - Lucite rotor shroud installed in the machine.

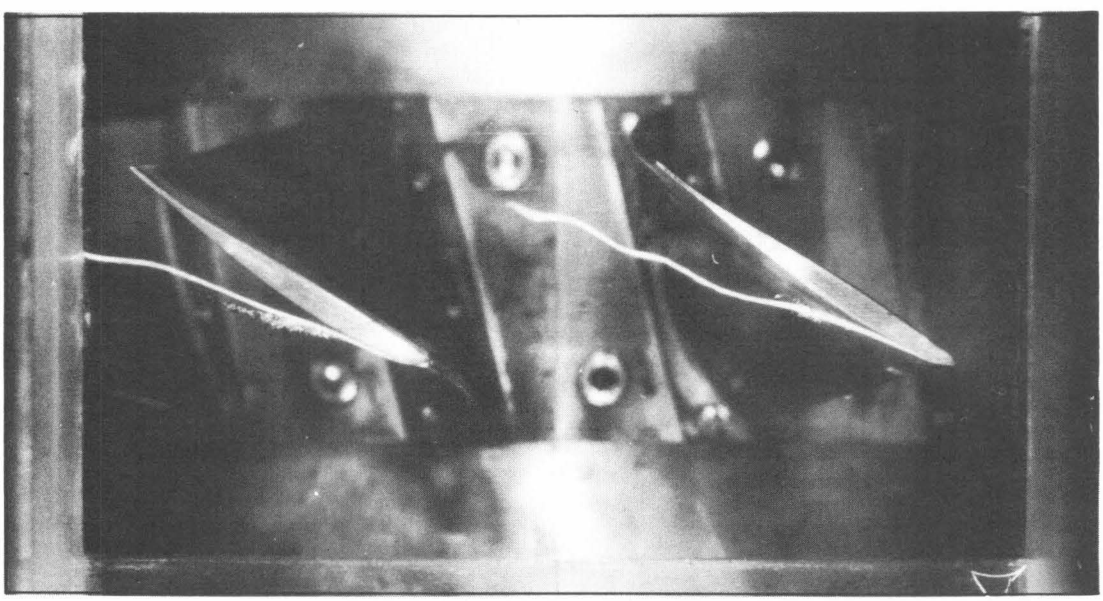

Fig. 8 - String attachment on blade, illustrated full size, showing cavitation at the design point, $K = 0.69$, $\lambda = 0.164$).

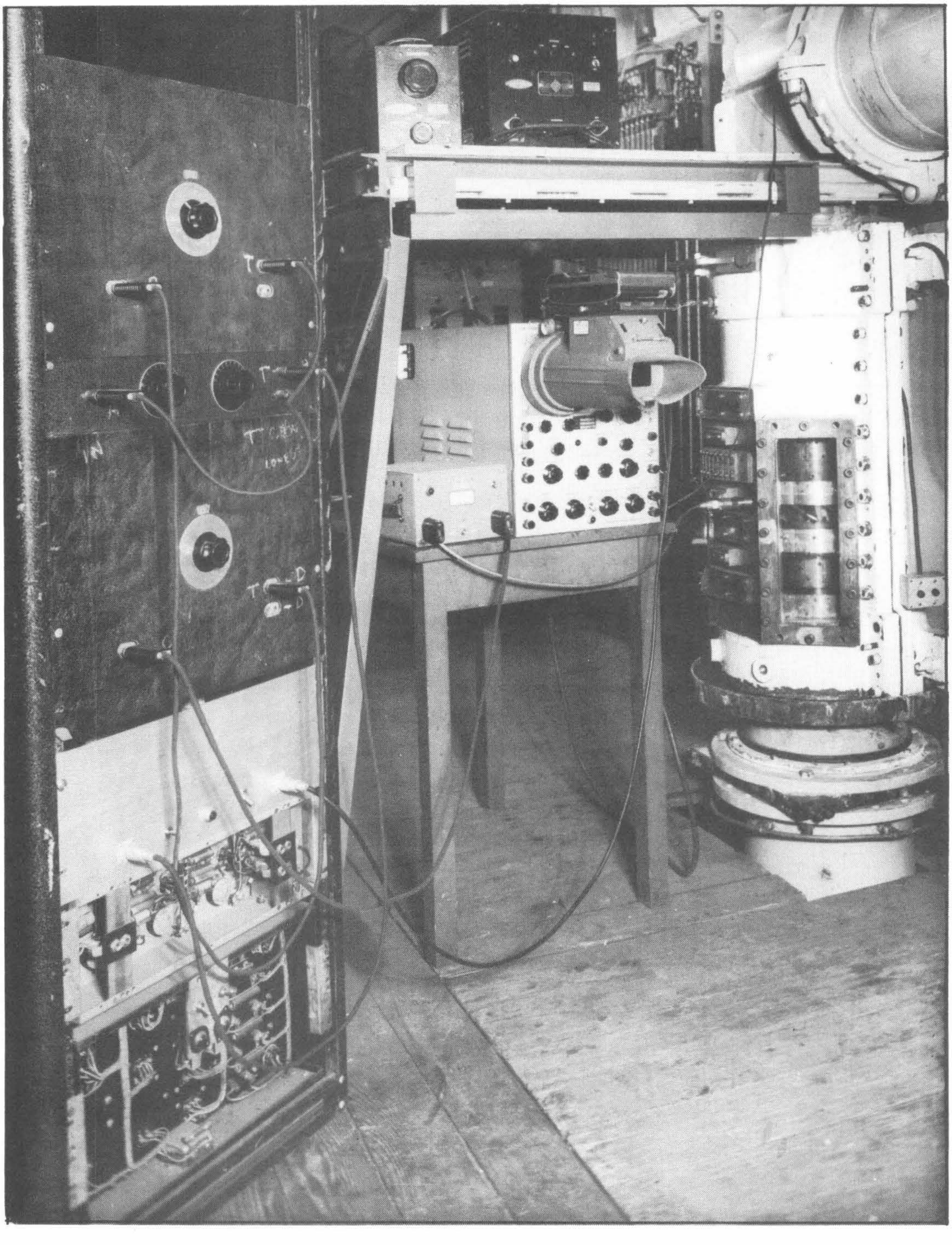

Fig. 9 - Sound detection apparatus showing hydrophone, oscilloscope and filtering equipment.

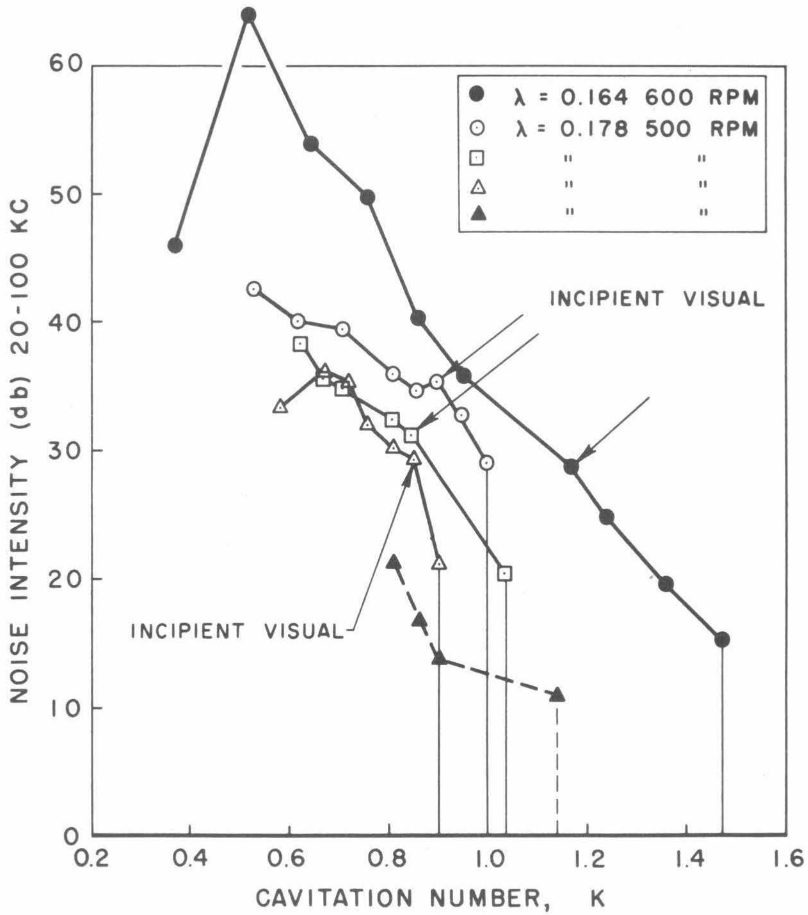

Fig. 10 - Noise level as a function of cavitation number for lucite feet attachments at the design rate of flow at two different speeds.

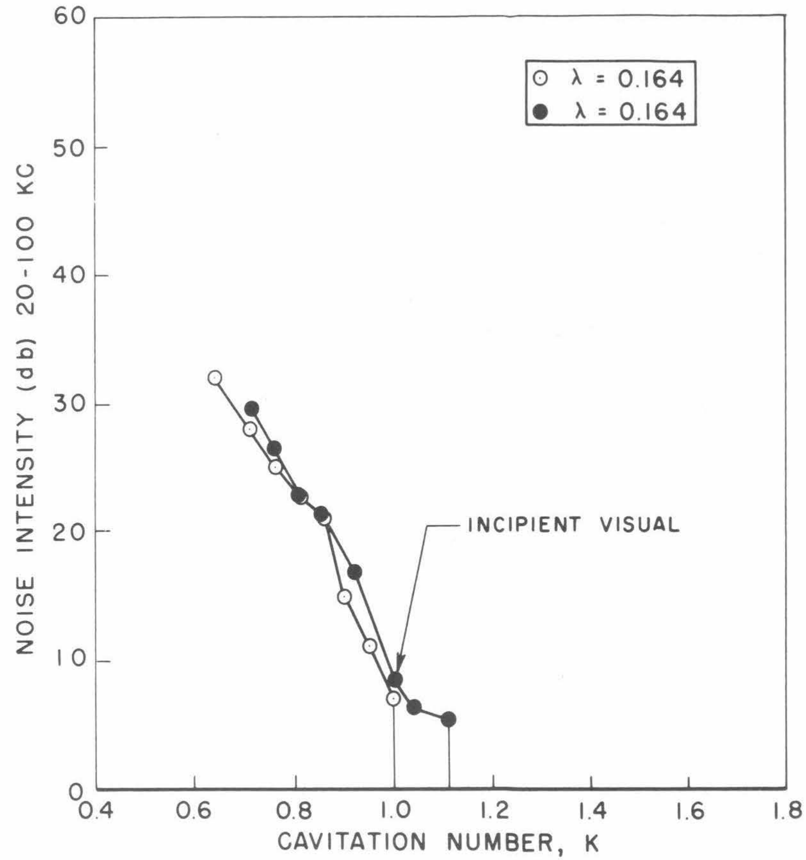

Fig. 11 - Noise level as a function of cavitation number for string attachments at the design rate of flow.

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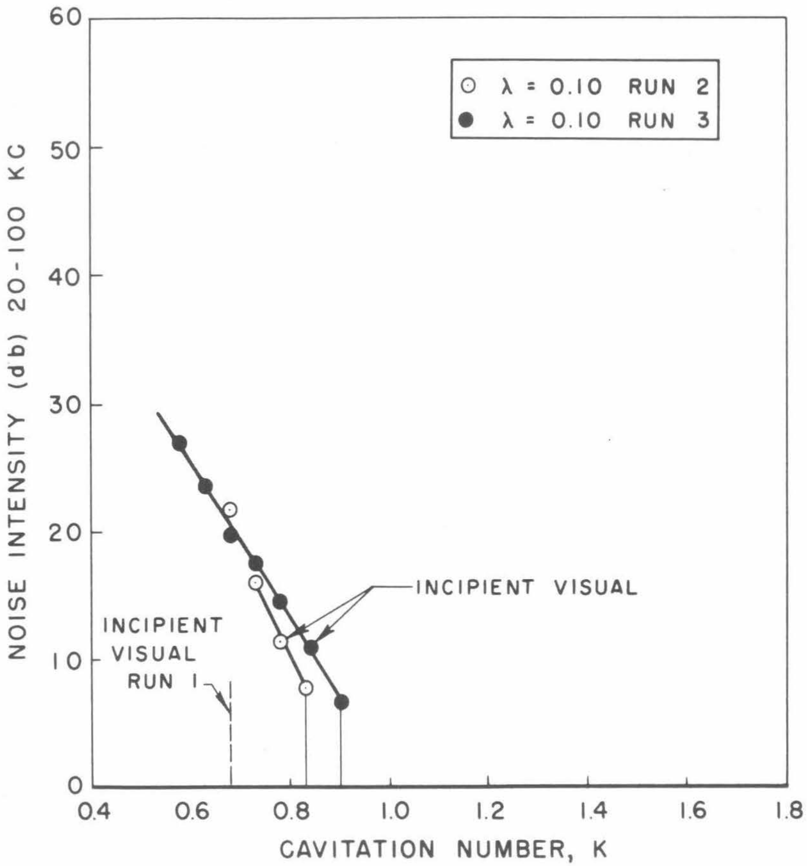

Fig. 12 - Noise level as a function of cavitation number for the lucite shroud at the design rate of flow.

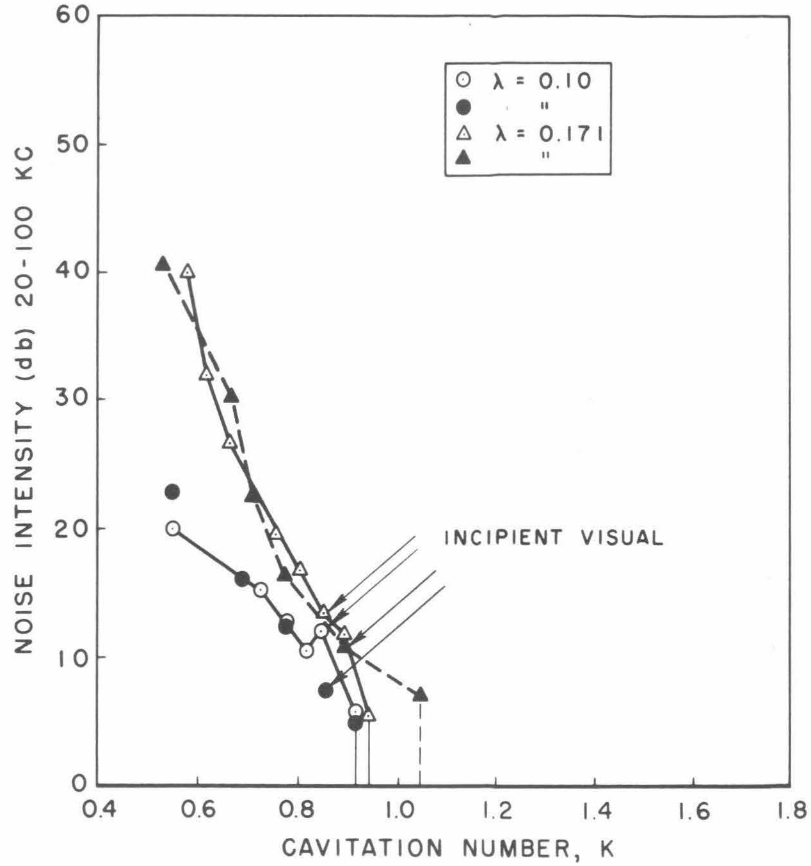

Fig. 13 - Noise level as a function of cavitation number for plain blades at the design rate of flow.

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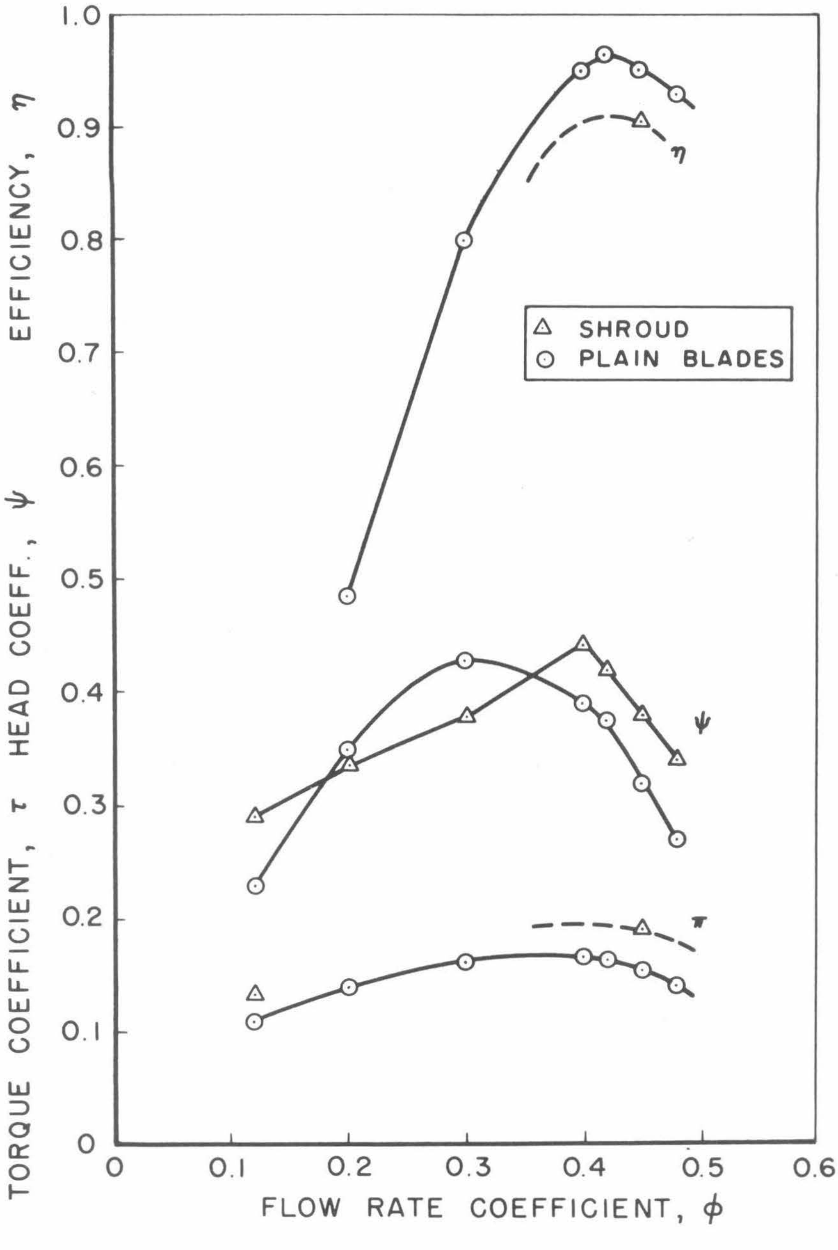

Fig. 14 - Comparison of shrouded and non-shrouded rotor performance.

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